NAVAL POSTGRADUATE SCHOOL

Monterey, California

CONDENSER HEAT TRANSFER AUGMENTATION

Progress Report

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1. BACKGROUND

The objective of this research program is to provide technical guidance and support to DTNSRDC, Code 2721 in an effort to develop a low weight, small volume condenser utilizing augmented titanium tubes. To accomplish this objective, the following specific tasks have been assigned for FY 81:

(a) Using advanced computer codes, analytically simulate a submarine main condenser and conduct a parametric analysis to determine the most promising condenser configurations for minimum weight and volume.

(b) Perform single tube and small bundle tests of selected, enhanced tubes using existing resources at the Naval Postgraduate School.

(c) Provide technical guidance in planning a large-scale experimental test program to verify the performance of augmented tube condensers in a marine environment.

(d) Continue to survey and assess advanced heat transfer enhancement techniques which may be applicable to marine condensers of the future.

This report summarizes the progress which has been made in each of the above areas and describes work to be accomplished during the balance of this fiscal year.

II. COMPUTER MODELING STUDIES

Progress to Date

During the first half of this fiscal year, efforts to model a submarine condenser have been centered around two codes: ORCON 2, a simplified one-dimensional condenser rating code, and CONDIP, a more general one-dimensional condenser analysis code which is capable of being coupled to numerical optimization routines. In addition, preliminary efforts have occurred to pursue the development of multi-dimensional codes. Each of these efforts is summarized below.

ORCON 2 Results. Tables 1 through 5 contain a variety of computer-generated results for the 688-class and the IPMP condenser. The condenser rating code used to obtain the various values reported here is an improved version of the ORCON code developed at the Oak Ridge National Laboratory. This program is by far the most comprehensive code for condenser performance that has been found to be available to the public. Some of the features and limitations of ORCON are summarized below:

(a) ORCON allows the calculation of radial flow of the steam/gas mixture across horizontal cooling tubes. There is no provision for flow in circumferential or axial directions.

(b) There is a means for correcting the condensate film

coefficient for the effects of condensate inundation. The method is based upon Eissenberg's[1] adaptations to the classical method of Nusselt. The correction only accounts for the distribution of condensate rain upon lower tubes in a gravity-dominated flow and does nothing else to modify the classic Nusselt model for laminar film condensation on an isolated tube.

(c) Non-condensible gases (up to four species) are taken into account by the use of the Reynolds flux model, as elaborated by Spalding, and an empirical correlation suggested by Eissenberg. The method appears to be similar to that employed elsewhere and represents the state-of-the-art in this rather uncertain area.

(d) ORCON is easily modified to include enhancement of the inner and/or outer film coefficients. If necessary, the enhancement may be calculated as a function of local conditions within the condenser.

(e) ORCON includes fouling effects only as a constant resistance term.

(f) All fluid properties are continuously variable and change with pressure and temperature throughout the condenser. The latter quantities are allowed to vary due to inlet

[1]Eissenberg, D. M., "An Investigation of the Variables Affecting Steam Condensation on the Outside of a Horizontal Tube Bundle," Ph.D. Thesis, U. Tennessee, Dec. 1972.

outlet, and tube-to-tube flow variations, as well as viscous losses, and the effects of condensation.

(g) The overall heat transfer coefficient is calculated on the basis of an "average" tube in each row of tubes. This is a basic limitation of the code that is somewhat relieved by the ability to calculate performance by sectors.

(h) In ORCON 1, vapor shear effects are not included. ORCON 2, used in this study, includes a modificiation to account for vapor shear effects. This modification is based upon the semi-empirical results of Fujii, et al.[1]

Table 1 summarizes the validation data obtained by ORCON 2 for the SSN 688 condenser. In examining the data, it is clear that ORCON 2 is well validated when compared to the specifications provided from NAVSEA. In a similar way, Table 2 lists the baseline data for the IPMP condenser and compares these to ORCON 2 results. Agreement, once again, is very good. Table 3 includes a wide variety of data for the IPMP condenser containing smooth or enhanced tubes and operating at a steam pressure of 6.5 inches Hg. Condenser length, bundle volume, and dry weight, as well as cooling water head loss and pumping power, are calculated

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[1]Fujii, T., "Vapor Shear and Condensate Inundation," Proceedings of the Workshop on Modern Developments in Marine Condensers, Naval Postgraduate School, Monterey, California, March 1980, pp. 193-223.

for three different tube diameters and cooling water flow rates. Table 3(a) provides data for the smooth tube case (note that outside and inside enhancement factors F_o and F_i are set equal to unity) whereas Tables 3(b) through 3(f) are for various enhanced tubes. The enhancement factors for these tubes were those specified by NSRDC[1] (Table I, p. 4 of enclosure (4) to CO letter 2721: RWK 3900). Appendices A, B, and C provide details on the enhanced tube friction factors as well as the equations used to calculate the head loss, pumping power, bundle volume, and dry weight.

In examining these tables, it is at first confusing to see that for the smooth tube, at a given cooling water flow rate, the required condenser length for 5/8 inch tubes is smaller than for either the 1/2 inch or 3/4 inch tube cases. A similar trend occurs for the YL1 and YL2 tubes which give minimum heat transfer enhancement on the inside. Apparently, this existence of a minimum length is due to the opposing

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[1]Note that the enhancement factors were assumed to remain constant as tube diameter changed. If the results of Fenner and Ciftci are compared, however, it appears that tube diameter may influence the values of F_o and F_i. Further work may be needed in this area. Wither's correlation for inside enhancement shows a dependence upon p/D_i and e/D_i. Similar conclusions apply to the YIM tubes. All of the results obtained to date for a given enhanced tube are for the same p/D_i and e/D_i. Hence, as tube diameter has changed, it has tacitly been assumed that both the pitch p and the groove depth e have changed correspondingly to keep p/D_i, e/D_i constant.

effects of tube diameter upon condenser surface area A and overall heat transfer coefficient U. As tube diameter increases, the surface area increases but the overall heat transfer coefficient decreases. (For this situation both h_i and h_o decrease with an increase in tube diameter). The optimum diameter depends upon a balance between the thermal resistances on the outside, the inside, and within the wall. For the smooth tube and the YL1 tube, this optimum diameter occurs between 1/2 inch and 5/8 inch. On the other hand, for the other enhanced tubes with significant enhancement on the inside, the optimum tube diameter shifts to a value greater than 3/4 inch so that the data in Tables 3(b), (c), and (d) show tube length to be monotonically decreasing with an increase in tube diameter.

Another general observation in Table 3 is that as the flow rate is reduced to 6000 GPM, the required tube length increases dramatically. It is clear, therefore, that the 6000 GPM case is not a viable option for submarine condensers at these heat load values.

In comparing the performance of the enhanced tubes to the smooth tube, the best heat transfer performance is given by tube YM2 which includes good enhancement on both the inside and the outside. For the 5/8 inch tube with a flow rate of 7900 GPM, the YM2 tube gives a tube length of 12.17 feet, a reduction of 36 percent from the IPMP baseline value of

18.95 feet. However, because of its larger friction factor, this tube gives a bundle head loss of 68.7 feet compared to only 12.93 feet for the baseline design. If the flow rate is increased to 10,000 GPM, the YM2 tube requires a length of only 9.78 feet, a 48 percent reduction. The head loss, however, increases to 88.4 feet which is near the upper constraint of 100 feet for this flow rate. Since the bundle head loss does not include the losses due to flow in the water box or to inlet and exit flow into and out of the tubes, this design condition may not be a viable alternative.

The minimum volume occurs for the 1/2 inch diameter YM2 tube operating at a flow rate of 10,000 GPM. A 64 percent reduction in volume occurs for this case at the expense of a severe head loss of 295 feet. This significant reduction in volume is due to the reduced bundle diameter as well as the reduced length. As shown in Table A-1, the baseline bundle diameter is 4.15 feet whereas the 1/2 inch tube <u>with the same number of tubes</u> allows a smaller bundle diamter of only 3.32 feet. An interesting extension of this case would allow the number of tubes to increase in order to preserve the bundle diameter near the baseline case. In so doing, the flow rate per tube is reduced and a sizable reduction in head loss would occur. These trends are illustrated in Table 4 which shows some preliminary results for the YM2 tube with a varying number of tubes. For the 1/2 inch tube,

2446 tubes per pass would provide a bundle diameter of 4.11 feet, a value very near the baseline case. With the addition of extra tubes, the required tube length is reduced to 7.91 feet, a 58 percent reduction of the baseline length. The bundle volume is also reduced by 59 percent. More importantly, the required cooling water velocity is reduced from 12.59 ft/sec to 8.47 ft/sec which gives a lower head loss of only 98.9 feet. If the number of tubes is increased to 3846, which gives a bundle diameter roughly 1 foot larger than the baseline case, further reductions occur. Perhaps the 5/8 inch tube case should be studied with varying numbers of tubes in order to compare performance to the other tube diameter cases. Other S/D ratios may also prove useful in this comparison.

Table 5 lists the results for the 5/8 inch diameter tubes operating at a reduced steam pressure of only 3.5 inches Hg. At a flow rate of 7900 GPM, the ORCON 2 results show that the required tube length is excessive due to a low LMTD caused by low steam temperatures. At 10,000 GPM a smooth tube requires a 31 foot long condenser whereas the YM2 tube requires a length of only 21.3 feet, a reduction of 31 percent. It may be more relevant in this situation to ask what partial heat load could be transferred at 3.5 inches Hg for a condenser designed to transfer a full heat load at 6.5 inches Hg.

CONDIP Results. CONDIP (Condenser Design Improvement Code) has its origins in the previously discussed ORCON 1 code. The main feature of CONDIP, however, is the capability to employ modern numerical optimization in the automatic attainment of optimum designs within a specified constraint framework. For this reason, the form of CONDIP is significantly different from that of ORCON 1.

At the present time, CONDIP is in a testing and debugging mode of operation. The basic condenser analysis routine has been successfully coupled to and run with the CODES/CONMIN numerical optimization code.

CONDIP, in its present version, has the ability to manipulate some 15 design variables (such as tube length, tube spacing, cooling water velocity, enhancement factors, etc.) in the search for an optimum objective function (such as bundle volume, bundle weight, pumping power, etc.). In this search, constraints such as maximum allowable pumping power are maintained by the code in accordance with user requirements. When operative, the code will in effect conduct an automatic search for the optimum design conditions, eliminating unusable designs (such as some of those shown in Table 3, for instance).

Because of its extraordinary flexibility, CONDIP is vulnerable to a number of programming difficulties. When a candidate design is not feasible (one which has insufficient

cooling water flow, for instance) protections against impossible calculations, such as logarithms of negative numbers, must be built into the code. This is the area of difficulty that is currently delaying the advancement of the code to operational status.

Two-Dimensional Code Development. The use of a one-dimensional code for condenser rating and analysis presupposes that the condenser design is one in which undesirable steam "meandering" is successfully avoided. That is, the steam does in fact follow the path invoked by the one-dimensional assumption. In ORCON 2 and CONDIP this path is radial.

The existence of design inadequacies associated with steam maldistribution and/or poor bundle layout will lead to circumferential and axial variations within the condenser. A major effect of such variations is the occurrence of trapped air pockets within the tube bundle. This and other effects cannot be predicted by an analysis that is primarily one-dimensional.

The extension of present capabilities to two- and three-dimensional steam flows constitutes a major computational expansion. As such, it is hoped that considerable benefits will be derived from existing technology in the general area of computational fluid dynamics (CFD). A number of contacts have been made with researchers in this are of endeavor and

some encouraging indications have been forthcoming.

In particular, communications with CHAM (Concentration, Heat, and Mass Transfer, Inc.), a London based CFD firm, indicate their confidence that appropriate codes can be developed with relative ease. Development of this connection is continuing.

Work to be Accomplished.

ORCON 2. A variety of parametric studies could be performed using the ORCON 2 code. These studies will depend upon the interests and desires of the sponsor. Such studies could include, for example, the following specific cases:

(a) Repeat selected test cases which appear in Table 3 for a variable tube spacing.

(b) For the 5/8 inch diameter YM2 tube, vary the number of tubes at a constant flow rate of 10,000 GPM to study bundle diameter effects.

(c) For the YM2 tube, obtain a series of runs with variations in F_i and F_o in order to study the sensitivity of the code output to these variables. Such a result may be very important in projecting the experimental heat transfer research which is still required.

CONDIP. Continue to test and debug the present code to eliminate the difficulties referred to above. Once operable, CONDIP could be run to arrive at optimized design conditions.

Two-Dimensional Code Development. Correspondence with CHAM will be continued to further pursue the possibility of more general codes. Efforts should be made to seek NSRDC administrative approval and/or guidance to proceed with negotiations between NPS and CHAM. A possible visit by CHAM personnel to NSRDC will be encouraged. In the event these discussions lead to CHAM involvement, additional financial support in the neighborhood of 5-10K would be sought for FY 81.

III. SINGLE TUBE/SMALL BUNDLE TESTS

Progress to Date

During the initial part of this fiscal year, efforts were made to improve the steam flow distribution within the 3 foot long, 5 tube bundle test condenser.[1] Secondary flow of the steam was causing the condensate to move axially along the condenser tubes prior to dropping onto the next tube in the bundle. This behavior could not be controlled and led to erratic results on the effect of condensate inundation.

Consequently, a decision was made in November 1980 to redesign the test condenser to eliminate this difficulty. A new stainless steel condenser was therefore designed by LT Roger Morrison.[2] The features of this new condenser include a streamlined diffuser and exit plenum as well as a shortened tube length of 12 inches. Furthermore, the five active tubes are in an in-line rather than a staggered arrangement. Figure 1 shows a sketch of this new condenser while Figures 2 and 3 are photographs showing the condenser

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[1] Demirel, I., "The Effect of Condensate Inundation on Condensation Heat Transfer in Tube Bundles of Marine Condensers," M.S. Thesis, Naval Postgraduate School, December 1980.

[2] Morrison, R., "A Test Condenser to Measure Condensate Inundation Effects in a Tube Bundle," M.S. Thesis, Naval Postgraduate School, March 1981.

from two different perspectives. This condenser was operated by LT Morrison to obtain preliminary data and to check the accuracy of the instrumentation. During this operation, several problem areas were detected which are outlined below:

(a) During the inital tests, the five copper test tubes which were used showed patches of dropwise condensation along portions of their surfaces. A complete filmwise condition was never reached even though the tubes were treated chemically to provide a fully wetted surface. Most probably, the test section was being contaminated by the steam supply, by machining oils which were deposited within parts of the new apparatus, or by outgassing from sealing gaskets.

(b) The wall temperatures of the test tubes were measured by small metallic sheathed thermocouples which were soldered into grooves within the tube walls. These thermocouples had a sheath outside diameter of only 0.020 inches and, as a consequence, several of these thermocouples broke during assembly and cleaning. However, inasmuch as there were four thermocouples per tube, it was still possible to get a reasonable measurement of average tube wall temperature.

(c) In comparing the measured average heat transfer coefficient to the coefficient predicted by the Nusselt theory, it was found that the experimental value was approximately 5 percent higher than the theory. This agreement is very

good. However, when a fin correction factor was incorporated into the data reduction to account for conduction effects out each end of the copper test tubes, the experimental data fell 15-20 percent below the theory (even with the presence of some dropwise patches as noted earlier). This discrepancy implies that either the uncertainty of the measurements was too large or noncondensable gases were influencing the heat transfer.

Despite the difficulties encountered, it was possible to operate the condenser and compare condensate inundation effects to theoretical predictions. Figure 4 shows a comparison between the average of four experimental runs and the theoretical predictions of Nusselt, Chen, and Eissenberg. The trend of the experimental data is reasonable. It shows that the ratio of the average heat transfer coefficient for an in-line bundle of N tubes to that for only a single tube can be represented by:

$$\frac{\bar{h}_N}{h_1} = N^{-0.14}$$

The exponent of 0.14 lies between the usually quoted values of 0.07 to 0.25.

Work to be Accomplished

During the balance of this fiscal year, the following additional work will be accomplished:

(a) Repeat the smooth tube measurements to eliminate the measurement uncertainties outlined above.

(b) Test selected enhanced tubes to compare their performance to smooth tubes. Obtain data for both a single tube and for a five tube bundle.

(c) From this data arrive at enhancement factors on the inside and outside of these tubes and deduce an exponent on the condensate inundation effect.

IV. LARGE-SCALE TEST PROGRAM

Progress to Date

Little effort has been expended on this task other than to maintain contact between the U.S. Navy and those laboratories which habe experience in testing large tube bundles. These laboratories include Argonne National Laboratory, Oak Ridge National Laboratory and TVA in the U.S., the Ministry of Defence, the Central Electricity Research Laboratory, and GEC Power Engineering Ltd. in England.

Work to be Accomplished

Establish an overall test plan which clearly delineates the path to be followed from single tube tests up to large-scale bundle tests. Such a plan would identify the objectives of each test apparatus (and its test program), pointing out the expected information from each phase of the plan. Areas of overlap (if any) will be examined as well as unique capabilities to arrive at a unified plan which can be utilized to verify the developing computer codes. A trip to Argonne National Laboratory and other sites in the UK should prove extremely useful in this task. This test plan should be completed as soon as possible so that funds may be budgeted for FY 82.

V. ASSESSMENT OF ENHANCED HEAT TRANSFER METHODS

Progress to Date

A continuous search is being made to discover new methods for improving heat transfer within Naval condensers. This search has included the review of the open literature such as journals, theses, and technical reports. One such important publication for this purpose is Previews of Heat and Mass Transfer, a journal which alerts its readers to ongoing heat transfer research. A second important source of information is the patent literature. A recent report on patents involving augmentation of heat transfer has been published by Iowa State University for the Department of Energy.[1] This document was studied for patents which were applicable to marine condensers. As of this date 46 patents have been identified for further study.

Work to be Accomplished

1. The 46 patents noted above will be ordered through the NPS library. Upon receipt, they will be examined for suitability, catalogued as to enhancement technique, and analyzed for current or future use. Promising techniques will be brought to the attention of NSRDC.

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[1] Webb, R.L., Junkhan, G.H., and Bergles, A.E., "Bibliography of U.S. Patents on Augmentation of Convective Heat and Mass Transfer," Report HTL-25, Iowa State University, September 1980.

2. The literature will be continuously surveyed for new developments in enhanced heat transfer. Attendance at the National Heat Transfer Conference, where several sessions on enhanced heat transfer will occur, should prove enlightening as well.

3. For the remainder of FY 81, Dr. John W. Rose, Queen Mary College, University of London, will be an Adjunct Professor in the Department of Mechanical Engineering. Dr. Rose is well-known in the condensation heat transfer community and has particular expertise in dropwise versus filmwise condensation conditions. Dr. Rose is presently completing a study of enhancement techniques for condenser use which is being performed for the National Laboratory in Harwell, UK. His expertise and European contacts should provide additional useful information.

4. Drs. Rose and Marto will conduct some fundamental tests on enhancement of steam condensation using a small, clean, vacuum tight apparatus. The results of these tests will be used to provide further insight into the important mechanisms which exist on the outside of a condenser tube.

TABLE 1. BASELINE DATA AND ORCON2 VALIDATION

SSN688 CLASS

a. Comparison of Inputs

	SSN688[1] Specification	ORCON2 Input
Number of Tubes[2]	1646	1646
% Tubes in Cooler	6.0	6.0
Tube Spacing/Diameter	1.35	1.35
Tube OD, in.	0.625	0.625
Tube Wall Thickness, in.	0.065	0.065
Tube Material	70-30 CuNi	—
Tube Wall Conductivity, Btu/hr-ft-°F	—	17.0[8]
CW Velocity[3], ft/sec	8.0	8.0
CW Salt Concentration, % by weight	—	3.5
CW Inlet Temp., °F	66.1	66.1
Fouling Resistance,[4] hr-ft^2-°F/Btu	0.00033	0.00033
Steam Load[5], lb/hr	189,003	189,003
Air Content, lb/hr	—	33.7
Steam Inlet Saturation[6] Temperature, °F	133.73	133.73

TABLE 1. BASELINE DATA AND ORCON2 VALIDATION (continued)

SSN688 CLASS

b. Comparison of Outputs

	SSN688[1] Specification	ORCON2
Steam Condensed, lb/hr	224,000	223,990
Heat Duty, Million Btu/hr	192.3[7]	192.5
Total Cooling Surface, ft^2	10,509	10,190
Effective Tube Length, ft	19.51	18.92
Outlet Water Temp., °F	114.8	116.3
Final Temp. Diff., °F	18.87	17.43
Overall Heat Transfer, Coefficient, Btu/hr-ft^2-°F	479.3	510.1
LMTD, °F	38.2	37.04
Bundle OD, ft	4.4[9]	4.15
Void OD, ft	0.5[9]	0.48
Cooler Exit Steam Fraction, %	___	< 0.01
Condenser Steam Side Pressure Drop, in. Hg	___	0.13
Inlet/Outlet Steam Velocity, ft/sec	___	119.7/50.8

TABLE 1. BASELINE DATA AND ORCON2 VALIDATION (continued)

Notes:

1. NAVSEA 0946-LP-018-5010, Consolidated Equipment Manual for Main Condenser, After and Gland Steam Condenser and Air Ejectors (U), De Laval Turbine Inc., April, 1974. CONFIDENTIAL.

2. Number of tubes in each of two passes.

3. Given value corresponds to cooling water flow rate of 7,900 GPM.

4. Equivalent to 85% tube cleanliness factor.

5. Saturated steam based upon total steam flow of 224,000 lb_m/hr with 15.6% moisture content.

6. Given value is equivalent to 5.00 in. Hg absolute pressure at condenser inlet.

7. Based upon steam load divided by latent heat of condensation at condenser inlet conditions.

8. Given value is for 70-30 CuNi.

9. Values estimated from tube sheet layout drawing No. CW-3050 JP contained in the reference of note 1 above.

TABLE 2. BASELINE DATA AND ORCON2 PREDICTIONS

IPMP

a. Comparison of Inputs

	IPMP[1] Specification	ORCON[2] Input
Number of Tubes[3]	1646	1646
% Tubes in Cooler	——	6.0
Tube Spacing/Diameter	——	1.35
Tube OD, in.	0.625	0.625
Tube Wall Thickness, in.	0.035	0.035
Tube Material	Ti	——
Tube Wall Conductivity, Btu/hr-ft-°F	——	9.5[9]
CW Velocity[4], ft/sec	8.0	8.0
CW Salt Concentration, % by weight	——	3.5
CW Inlet Temp., °F	66.1	66.1
Fouling Resistance[5], hr-ft^2-°F/Btu	0.00033	0.00033
Steam Load[6], lb/hr	218,431	218,431
Air Content, lb/hr	——	33.7
Steam Inlet Saturation[7] Temperature, °F	143.89	143.89

TABLE 2. BASELINE DATA AND ORCON2 PREDICTIONS (continued)

b. Comparison of Outputs

	IPMP[1] Specification	ORCON2
Steam Condensed, lb/hr	257,000	256,996
Heat Duty, Million Btu/hr	221.0[8]	221.2
Total Cooling Surface, ft^2	10,509	10,209
Effective Tube Length, ft	19.51	18.95
Outlet Water Temp., °F	—	123.6
Final Temp. Diff., °F	—	19.3
Overall Heat Transfer Coefficient, Btu/hr-ft^2-°F	—	506.7
LMTD, °F	—	42.8
Bundle OD, ft	—	4.15
Void OD, ft	—	0.48
Cooler Exit Steam Fraction, %	—	< 0.01
Condenser Steam Side Pressure Prop, in. Hg	—	0.14
Inlet/Outlet Steam Velocity, ft/sec	—	108.0/46.2

TABLE 2. BASELINE DATA AND ORCON2 PREDICTIONS

IPMP

Notes:

1. SSN IMPROVED PERFORMANCE MACHINERY PROGRAM, Engine Room Cooling Assessment, Task M50-1-1, General Dynamics Electric Boat Division, Contract No. N00024-78-C-2865, Dec. 1978. CONFIDENTIAL.

2. Where specific values could not be determined, values for SSN688 Baseline were used.

3. Number of tubes in each of two passes.

4. Given value corresponds to cooling water flow rate of 7,900 GPM.

5. Equivalent to 85% tube cleanliness factor.

6. Saturated steam based upon total steam flow of 257,000 lb_m/hr with 15% moisture content.

7. Given value is equivalent to 6.5 in. Hg absolute pressure at condenser inlet.

8. Based upon steam load divided by latent heat of condensation at condenser inlet conditions.

9. Given value is for titanium.

TABLE 3. ORCON2 PERFORMANCE PREDICTIONS

SMOOTH AND ENHANCED IPMP TUBES

a. Smooth Tubes, $F_o = F_i = 1.000$

Total CW Flow GPM	Tube OD/ Wall IN.	Bundle Effective Length,* ft.	Bundle Volume,* ft^3	Bundle Weight,* lb.	Bundle Head Loss,* ft.	Pumping Power,* Horsepower
10,000	0.750/0.049	14.98	292	10,420	7.17	18.5
	0.625/0.035	14.74	199	6160	15.2	39.1
	0.500/0.028	15.27	132	4080	45.5	117
7,900	0.750/0.049	19.36	377	13,460	6.14	12.6
	0.625/0.035	18.95	256	7,920	12.93	26.4
	0.500/0.028	19.37	168	5,180	38.1	77.9
6,000	0.750/0.049	45.56	887	31,600	8.94	13.9
	0.625/0.035	43.59	590	18,220	18.4	28.5
	0.500/0.028	44.73	387	11,960	54.4	84.4

* Values based upon effective length only. No allowances are included for support length and inlet/exit losses.

TABLE 3. ORCON2 PERFORMANCE PREDICTIONS

SMOOTH AND ENHANCED IPMP TUBES (Continued)

b. Korodense tubes (KM), $F_i = 2.51$, $F_o = 1.14$

Total CW Flow GPM	Tube OD/ Wall in.	Bundle Effective Length,* ft.	Bundle Volume,* ft^3	Bundle Weight,* lb.	Bundle Head Loss,* ft.	Pumping Power,* Horsepower
10,000	0.750/0.049	10.88	212	7560	22.8	58.7
	0.625/0.035	11.09	150	4634	49.8	128
	0.500/0.028	12.16	105	3240	158	408
7,900	0.750/0.049	13.63	265	9480	18.9	38.6
	0.625/0.035	13.78	186	5760	41.1	83.9
	0.500/0.028	14.99	130	4000	129	263
6,000	0.750/0.049	30.39	591	21,080	26.0	40.4
	0.625/0.035	30.46	412	12,732	56.1	87.1
	0.500/0.028	32.89	284	8,800	175	271

* Values based upon effective length only. No allowances are included for support length and inlet/exit losses.

TABLE 3. ORCON2 PERFORMANCE PREDICTIONS

SMOOTH AND ENHANCED IPMP TUBES (Continued)

c. Yorkshire maximum heat transfer tubes (YM1). $F_i = 2.70$, $F_o = 1.38$.

Total CW Flow GPM	Tube OD/ Wall in.	Bundle Effective Length,* ft.	Bundle Volume,* ft^3	Bundle Weight,* lb.	Bundle Head Loss,* ft.	Pumping Power,* Horsepower
10,000	0.750/0.049	10.27	200	7140	41.5	107
	0.625/0.035	10.46	141	4380	94.6	244
	0.500/0.028	11.47	99	3060	316	816
7,900	0.750/0.049	12.85	250	8940	32.4	66.2
	0.625/0.035	13.01	176	5440	73.5	150
	0.500/0.028	14.14	123	3780	244	498
6,000	0.750/0.049	28.47	554	19,740	41.4	64.3
	0.625/0.035	28.67	388	11,980	93.3	145
	0.500/0.028	30.96	268	8,280	307	477

* Values based upon effective length only. No allowances are included for support length and inlet/exit losses.

TABLE 3. ORCON2 PERFORMANCE PREDICTIONS

SMOOTH AND ENHANCED IPMP TUBES (Continued)

d. Yorkshire maximum heat transfer tubes with external enhancement. (YM2). $F_i = 2.70$, $F_o = 1.94$.

Total CW Flow GPM	Tube OD/ Wall in.	Bundle Effective Length,* ft.	Bundle Volume,* ft^3	Bundle Weight,* lb.	Bundle Head Loss,* ft.	Pumping Power,* Horsepower
10,000	0.750/0.049	9.68	188	6740	39.1	101
	0.625/0.035	9.78	132	4160	88.4	228
	0.500/0.028	10.70	92.6	2860	295	764
7,900	0.750/0.049	12.15	237	8460	30.6	62.7
	0.625/0.035	12.17	165	5100	68.7	140
	0.500/0.028	13.20	114	3540	227	465
6,000	0.750/0.049	27.18	529	18,908	39.5	61.4
	0.625/0.035	27.04	366	11,300	88.0	137
	0.500/0.028	29.03	251	7,780	288	447

* Values based upon effective length only. No allowances are included for support length and inlet/exit losses.

TABLE 3. ORCON2 PERFORMANCE PREDICTIONS

SMOOTH AND ENHANCED IPMP TUBES (Continued)

e. Yorkshire low pressure drop tubes (YL1). $F_i = 1.44$, $F_o = 1.35$.

Total CW Flow GPM	Tube OD/ Wall in.	Bundle Effective Length,* ft.	Bundle Volume,* ft^3	Bundle Weight,* lb.	Bundle Head Loss,* ft.	Pumping Power,* Horsepower
10,000	0.750/0.049	12.35	240	8600	11.1	28.5
	0.625/0.035	12.26	166	5120	23.8	61.2
	0.500/0.028	12.96	112	3460	72.9	188
7,900	0.750/0.049	15.77	307	10,980	9.25	18.9
	0.625/0.035	15.55	210	6500	19.7	40.4
	0.500/0.028	16.23	140	4340	60.0	122
6,000	0.750/0.049	36.62	705	25,200	12.9	20.2
	0.625/0.035	35.36	478	14,780	27.4	42.6
	0.500/0.028	36.24	314	9,700	81.8	127

* Values based upon effective length only. No allowances are included for support and inlet/exit losses.

TABLE 3. ORCON2 PERFORMANCE PREDICTIONS

SMOOTH AND ENHANCED IPMP TUBES (Continued)

f. Yorkshire low pressure drop tubes with external enhancement (Y12). $F_i = 1.44$, $F_o = 1.94$.

Total CW Flow GPM	Tube OD/ Wall in.	Bundle Effective Length,* ft.	Bundle Volume,* ft^3	Bundle Weight,* lb.	Bundle Head Loss,* ft.	Pumping Power,* Horsepower
10,000	0.750/0.049	11.71	228	8140	10.5	27.1
	0.625/0.035	11.52	156	4820	22.3	57.5
	0.500/0.028	12.13	99.7	3240	68.2	176
7,900	0.750/0.049	15.02	292	10,440	8.82	18.0
	0.625/0.035	14.68	198	6,140	18.7	38.0
	0.500/0.028	15.23	132	4,080	56.35	115
6,000	0.750/0.049	34.58	674	24,000	12.4	19.2
	0.625/0.035	33.54	454	14,020	26.0	40.4
	0.500/0.028	34.46	298	9,220	77.7	120

* Values based upon effective length only. No allowances are included for support length and inlet/exit losses.

TABLE 4. PRELIMINARY SURVEY OF TUBE BUNDLE DIAMETER EFFECTS

YM2 TUBES, 10,000 GPM CW FLOW, 6.5 in. Hg.

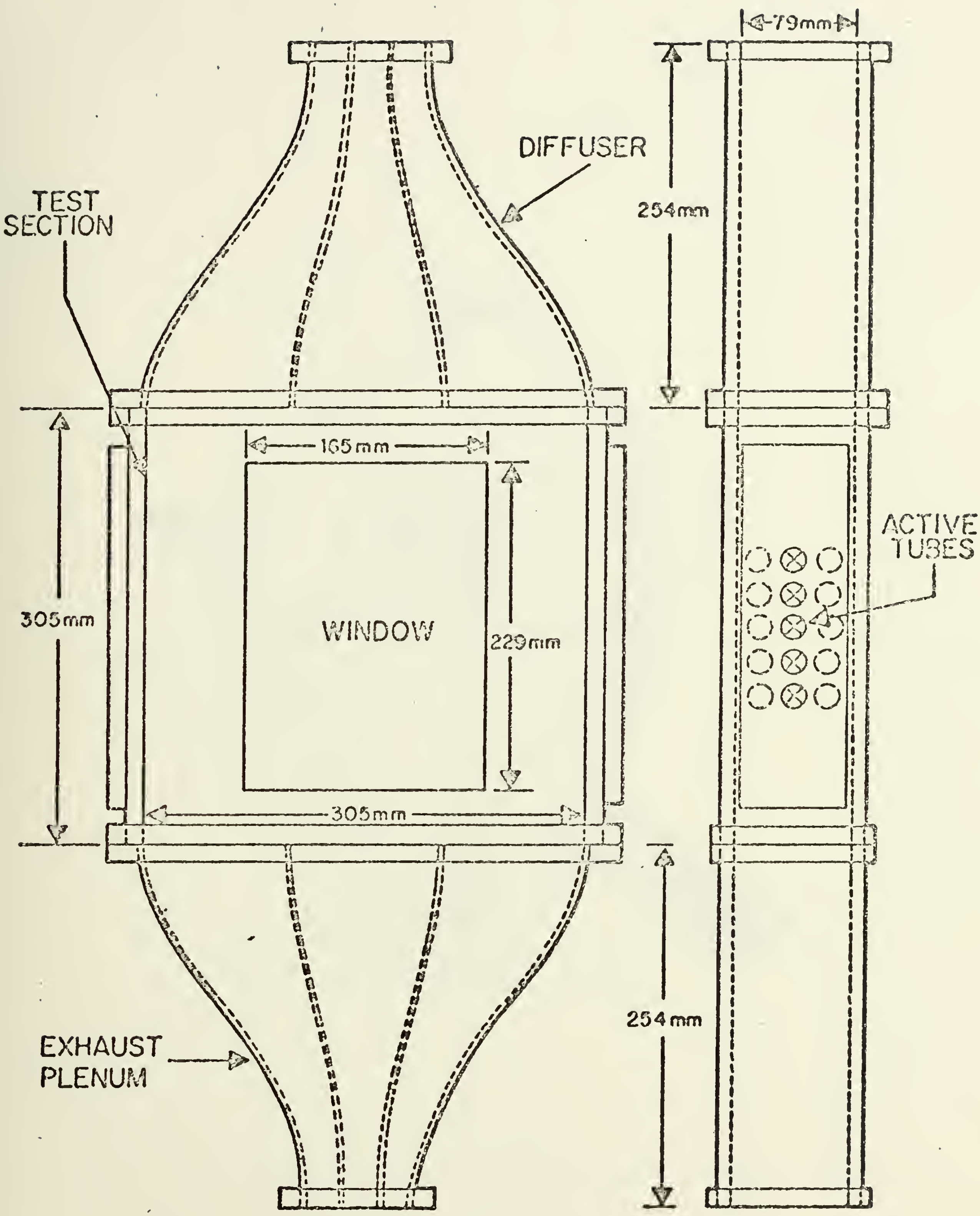
Tube OD, in.	Number of Tubes	CW Velocity, ft/sec.	Bundle OD, ft.	Tube Length, ft.	Bundle Volume, ft^3	Bundle Weight, lb.	Head Loss, ft.	Pumping Power, hp.
0.750	1646	5.8368	4.98	9.68	188	6740	39.1	101
	1746	5.5025	5.15	9.26	193	6830	33.3	86.1
	1846	5.2045	5.32	8.87	197	6920	28.5	73.8
	2434	3.9472	6.16	7.26	216	7470	13.4	34.7
0.500	1646	12.587	3.32	10.70	92.6	2860	295	764
	2446	8.4699	4.11	7.91	105	3150	98.9	256
	3846	5.3868	5.12	5.83	120	3650	29.5	76.2

TABLE 5. ORCON2 PREDICTIONS FOR SELECTED CONDITIONS

IPMP: 3.5" in. Hg steam inlet pressure, 5/8 - inch tubes

	Total CW Flow, GPM			
	7,900		10,000	
	Bundle Length ft.	Exit Fraction %	Bundle Length, ft	Exit Fraction %
Smooth tube	Max*	8.22	31.0	.01
Korodense	Max*	6.37	23.8	.01
Yorkshire YM1	Max*	6.19	22.6	.01
Yorkshire YM2	Max*	6.07	21.3	.01
Yorkshire YL1	Max*	6.84	26.2	.01
Yorkshire YL2	Max*	6.15	24.8	.01

* At the lower flow rate (7900 GPM) the calculation is terminated at a tubelength of 50 ft. (designated by Max). Due to low LMTD, (caused by low steam temperatures), heat transfer surface area approaches infinity for low exit fractions. Exit fractions shown for this condition are for bundle lengths of 50 feet.



NOTE: ALL COMPONENTS DRAWN TO SCALE

Figure 1: Sketch of the new 5 tube test condenser

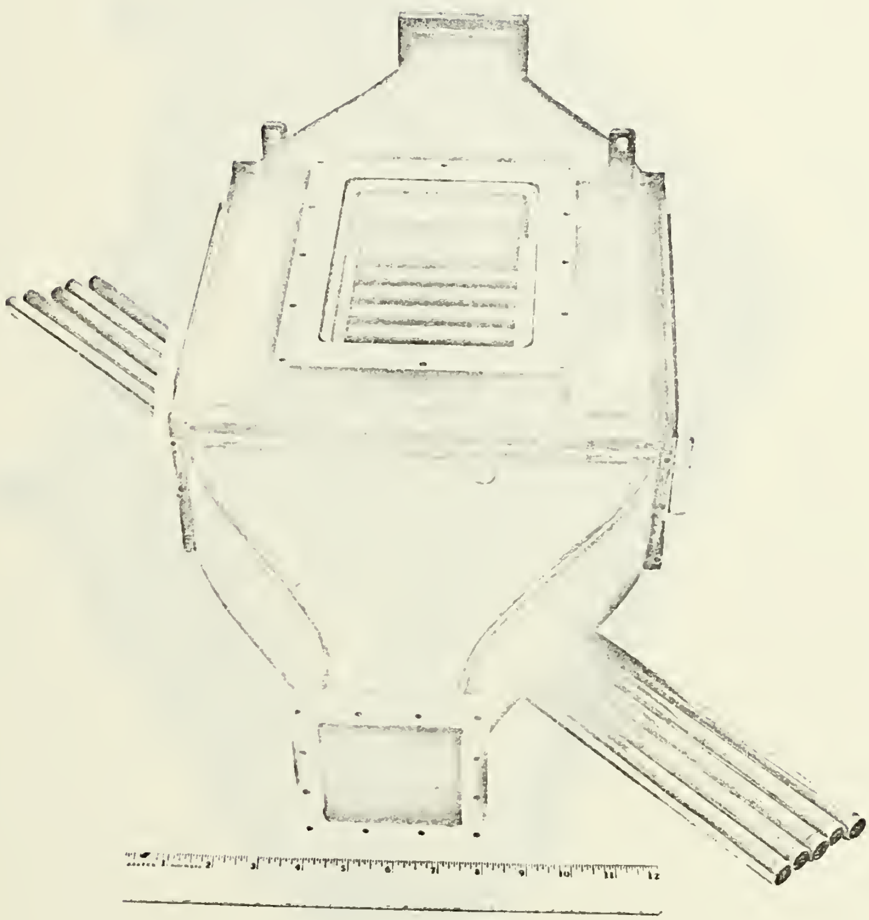


Figure 2. Photograph of the new test condenser showing inlet diffuser and viewing window

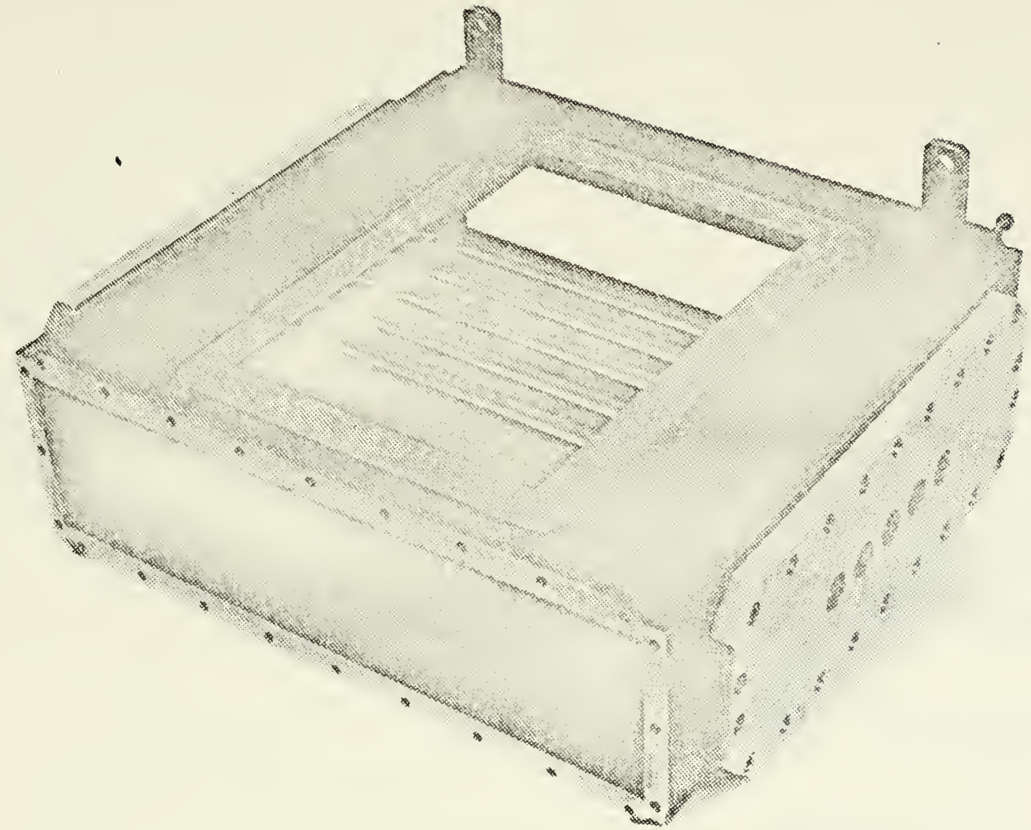


Figure 3: View of test tubes through window of test condenser

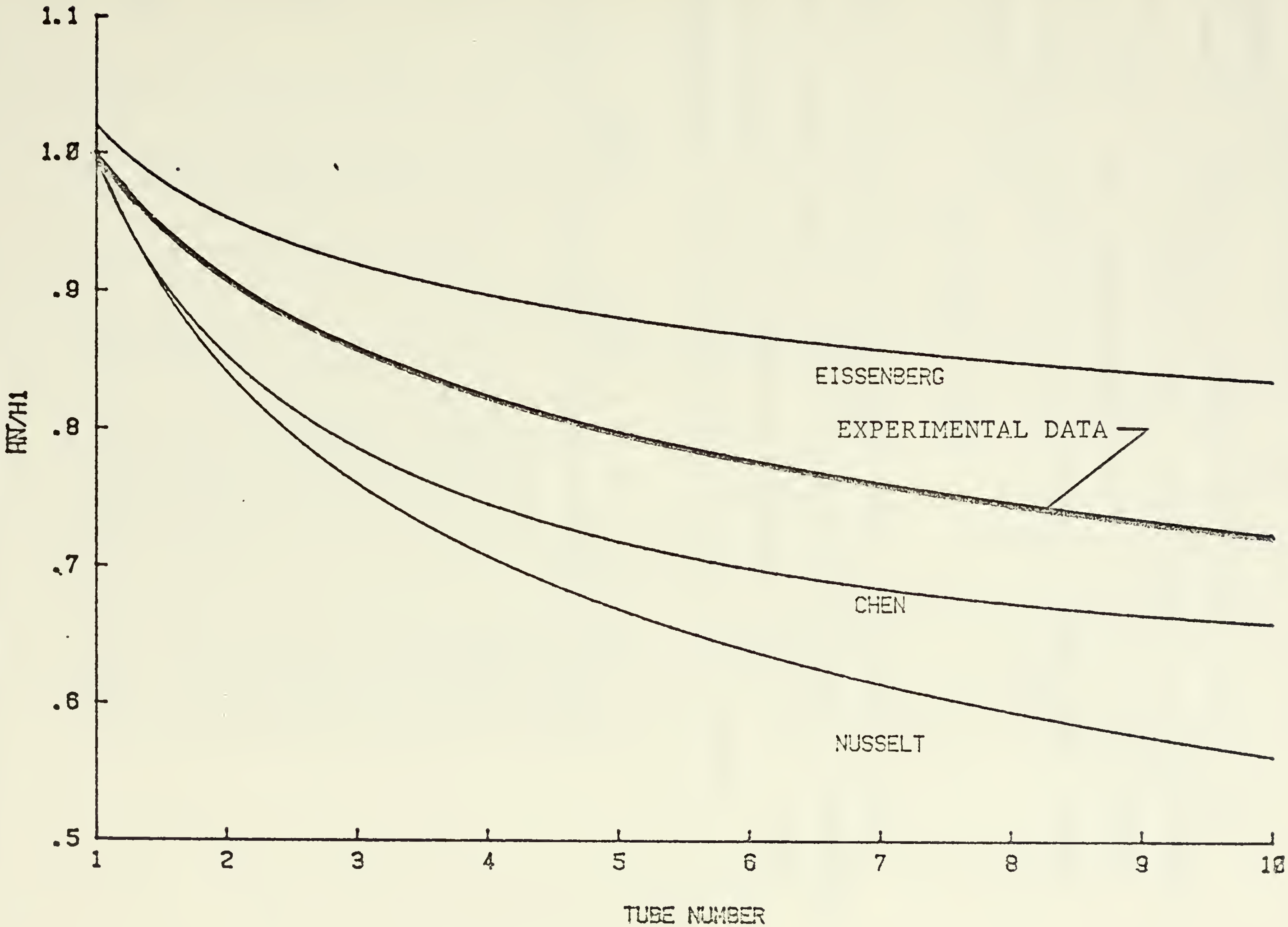


Figure 4: Comparison of Experimental Data to Tube Bundle Predictions.

Appendix A: Enhanced Tube Friction Factors

Korodense, K_m

A baseline value of f = 0.1096[1] is used for the 5/8-inch Korodense K_m tube at 7900 GPM (the corresponding Reynolds number is 2.53×10^4). The corresponding smooth-tube friction factor at this Reynolds number is fs = 0.0251 (see Table A-1), giving the following ratio:

$$F_f = \frac{f}{fs} = 4.366$$

This ratio is assumed to be constant over the Reynolds number range of interest and the smooth tube friction factor is assumed to vary according to the Blasius correlation ($f \propto Re^{-1/4}$). With these assumptions the following table may be constructed.

Q GPM	Tube OD	Re $\times 10^{-4}$	fs	$f(K_m)$
10,000	.750	2.73	0.0246	0.1074
	.625	3.20	0.0236	0.1031
	.500	4.01	0.0224	0.0978
7,900	.750	2.16	0.0261	0.1140
	.625	2.53	0.0251	0.1096
	.500	3.16	0.0237	0.1035
6,000	.750	1.64	0.0280	0.1222
	.625	1.92	0.0269	0.1174
	.500	2.40	0.0254	0.1110

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[1] This value is four times larger than the value provided in Table I, p. 4 of enclosure (4) to NSRDC letter.

TABLE A - 1. BASELINE VALUES FOR SMOOTH TUBE FRICTION FACTORS AND BUNDLE DIMENSIONS

TOTAL CW FLOW GPM (1646 TUBES)	D/d TUBE OD/ID IN.	v CW VELOCITY FT/SEC	Re* REYNOLDS NUMBER	f** FRICTION FACTOR (SMOOTH)
10,000	.750/.652	5.8368	2.73E04	0.0246
	.625/.555	8.0554	3.20E04	0.0236
	.500/.444	12.5866	4.01E04	0.0224
7,900	.750/.652	4.6111	2.16E04	0.0261
	.625/.555	6.3638	2.53E04	0.0251
	.500/.444	9.9434	3.16E04	0.0237
6,000	.750/.652	3.5021	1.64E04	0.0280
	.625/.555	4.8332	1.92E04	0.0269
	.500/.444	7.5519	2.40E04	0.0254

TUBE OD IN.	BUNDLE DIAMETER FT.
.750	4.98
.625	4.15
.500	3.32

* $Re = \frac{Vd}{\nu}$, $\nu = 1.1625 \times 10^{-5}$

** $f = \frac{0.3164}{Re^{\frac{1}{4}}}$, $H = f \, \frac{l}{d} \, \frac{V^2}{2g}$

Yorkshire maximum heat transfer, Y_m

Tubes having the following geometry are assumed:

$$P/d_i = 0.125, \quad e/d_i = 0.0215$$

The friction factor is given by:[1]

$$f = fr\left(\frac{Re}{10^5}\right)^{C_6}$$

where fr and C_6 are provided by manufacturer's data for a particular tube. In this case,

$$fr = 0.2076$$

$$C_6 = -0.013$$

so that

$$f = 0.2076\left(\frac{Re}{10^5}\right)^{-.013}$$

The dependency on Reynolds number is insignificant in the range of interest so that

$$f = 0.2076 \quad (Y_m)$$

is taken constant for this tube.

Yorkshire low pressure drop, Y_L

Tubes having the following geometry are assumed:

$$P/d_i = 0.250, \quad e/d_i = 0.013$$

and, from manufacturer's data:

$$f = 0.035\left(\frac{Re}{10^5}\right)^{-0.21}$$

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[1]From Yorkshire Imperials Metals, Ltd. Technical Memorandum No. 3.

From this relationship the following table is constructed:

Q GPM	Tube OD	Re $\times 10^{-4}$	$f(Y_L)$
10,000	0.750	2.73	0.0460
	0.625	3.20	0.0444
	0.500	4.01	0.0424
7,900	0.750	2.16	0.0483
	0.625	2.53	0.0467
	0.500	3.16	0.0456
6,000	0.750	1.64	0.0511
	0.625	1.92	0.0495
	0.500	2.40	0.0471

Appendix B: Head loss calculations

The friction factors given in these developments are defined by

$$f = \frac{H}{\frac{L}{d_i}\frac{V^2}{2g}}$$

where H is the frictional head loss (ft)

V is the cooling water velocity (ft/sec)

g is the gravitational acceleration (ft/sec^2)

In calculating the head loss requirements given in Table 3, only the effective tube lengths are taken into account. The effects of waterbox inlet and outlet losses may be accounted for by adding H_{WB} where

$$H_{WB} = 1.5 + 0.42\ (V_{WB} - 8)$$

where V_{WB} is the waterbox velocity in feet per second. Likewise, tube end losses, H_E, may be approximated by

$$H_E = 1.2 + 0.6\ (V_t - 6)$$

where V_t is the tube water velocity in feet per second. Finally, it is customary to add 1.75 ft in equivalent tube length to account for non-active (tube extensions and support) lengths. For example, consider tube YM2 with 0.750 - in. OD and at 10,000 GPM (Table 3d). Assume V_{WB} = 8 ft/sec. For this case:

$$V_t = 5.8358 \text{ ft/sec.}$$

$$f = 0.2076$$

$$H_{WB} = 1.5 \text{ ft.}$$

$$H_E = 1.10 \text{ ft.}$$

$$L_E = 9.68 + 1.75 = 11.43 \text{ ft.}$$

$$H_{TOTAL} = 1.5 + 1.1 + \frac{0.2076\ (5.8368)^2}{2(32.2)} \left(\frac{11.43}{\frac{0.652}{12}}\right) \qquad (2)$$

$$H_{TOTAL} = 1.5 + 1.1 + 46.2 = 48.8 \text{ ft.}$$

Which should be compared with 39.1 ft for the basic effective tube length.

Pumping power in horsepower may be calculated by

$$P_p = 2.567 \times 10^{-4}\ QH$$

where Q = bundle flow, GPM

H = head loss, feet.

Appendix C: Calculation of Bundle Volume and Bundle Dry Weight

The bundle volume shown in the tables is the volume occupied by the effective length of the tubes.

$$V_B = \frac{\pi D_B^2}{4} L$$

where D_B = bundle diameter

L = effective tube length.

The bundle dry weight is the weight of the effective length of the tubes.

$$W_B = [\pi N t_w (do - t_w) L] \, w$$

where N = total number of tubes = twice the number of tubes per pass

t_w = tube wall thickness

do = tube outside diameter

w = tube material specific weight (282 lb/ft^3 for titanium).